A practical method for estimating dynamic forces that cause machinery vibration

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Vibration is one of the most significant variables affecting the life of rotating machinery. However, vibration is only the manifestation of dynamic forces. It is force which damages machines and can lead to fatigue-generated failures. If the forces that act on a machine are known, one can estimate their effect on the machine's operating lifetime. For example, if the magnitude of shaking forces applied to rolling element bearings is known, the machine's remaining life can be estimated. Such an understanding can also help reduce the risk of catastrophic failure that can arise if dynamic forces have not been accurately assessed.

Risk management is important to users of equipment that handle significant quantities of hazardous materials, especially with the new OSHA 1910.119 guidelines. Most of these users now rank risk management higher than all other concerns. However, reliability and risk are intimately connected. Generally, if plant-wide reliability is improved, risk is reduced. Therefore, machinery users should continue to attack equipment problems in order to improve reliability.

Industry-accepted criteria, such as the Rathbone chart, have been used in the past to set velocity limits on general-purpose equipment. This chart was designed for machinery with oil-film bearings, where shaft motion is controlled by the stiffness of the oil wedge supporting the rotor. Therefore, the charts assume a 2:1 to 3:1 ratio of shaft to bearing housing vibration.

For equipment with this ratio, the charts are a qualitative method of estimating vibration severity. However, in the majority of general-purpose machinery, this ratio is not two or three to one, but very close to unity. These machines use rolling element bearings, which couple the shaft rigidly to the bearing housing support structure. In a rolling element bearing system, the stiffness of the rotor support system is controlled primarily by the bearing housing stiffness. To understand this relationship, we must know the dynamic stiffness function for the frequency range of interest.

To investigate the relationship between support stiffness and dynamic shaking forces, we selected a group of process pumps for testing. Our goal was to develop a methodology to:

- Reduce the risk from rotor-transmitted shaking forces.
- Lengthen the operational life of a machine by setting vibration limits based on its dynamic stiffness characteristics, such as bearing dynamic load capacity.

The concept

Applying a shaking force to a bearing housing and measuring its response, when the shaking force's amplitude and phase are known, provides enough information to calculate the bearing housing's Direct Dynamic Stiffness (DDS) and Quadrature Dynamic Stiffness (QDS). Sweeping the shaking force

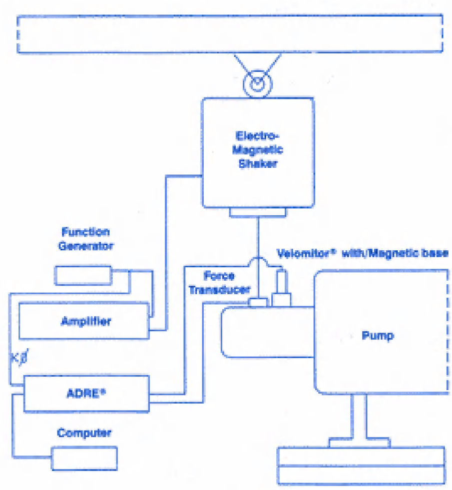


Figure 1
Test equipment layout

through a frequency range provides enough information to draw DDS and QDS curves for that range. Using this information, we can approximate the force transmitted through the bearing.

Dynamic Stiffness definitions

Complex Dynamic Stiffness (CDS)

The stiffness vector for a certain operating speed that is determined by dividing the input force vector by the response vector.

Direct Dynamic Stiffness (DDS)

The component of dynamic stiffness that acts collinearly to the input force vector. It is a function of the cosine of the angle between force and response.

Quadrature Dynamic Stiffness (QDS)

The component of dynamic stiffness that acts perpendicular to the input force vector. It is a function of the sine of the angle between force and response. Quadrature dynamic stiffness is a significant component in fluid film bearings because of the tangential forces created by the oil wedge that supports the shaft. Testing to date indicates that the QDS component exists in rolling element bearings but, at frequencies below the first balance resonance frequency, is usually a considerably smaller component than the DDS.

For nonsynchronous (swept frequency) perturbation, the Direct Dynamic Stiffness [Ref. 1] is defined as:

$$\mathrm{DDS} = \frac{F\cos(\alpha)}{A}$$

The Quadrature Dynamic Stiffness [Ref. 2] is defined as:

$$\mathrm{QDS} = \frac{-F\sin(\alpha)}{A}$$

where:

F = the applied force amplitude

α = the phase angle difference of the applied force and the response due to the applied force

A = the amplitude of the response due to the applied force

Consider general-purpose machines with low housing to rotor mass ratios. Force losses through the housing are negligible and the QDS component is significantly smaller than the DDS component. Most of these machines are mounted so rigidly that they operate well below any housing resonance, therefore they are spring stiffness dominated. Additionally, the rolling element bearing's high spring stiffness makes the machine behave as a rigid body. Therefore, given the DDS graph for a certain location, we can calculate the force, at that location, that has not been absorbed by the local housing mass. We calculate the force by multiplying the DDS by the vibration displacement value measured by the housing transducer. Assuming that all housing forces are generated by the rotor, we can approximate the force transmitted through the bearing at that point by the following equation:

$$F = KA$$

where

F = Apparent dynamic force at the frequency of interest

K = DDS at the frequency of interest

A = Response amplitude at the frequency of interest ►

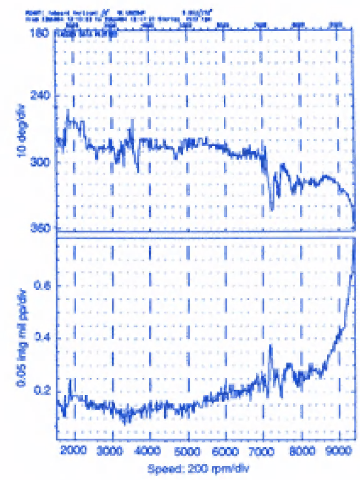


Figure 2
Bode plot during a typical calibration

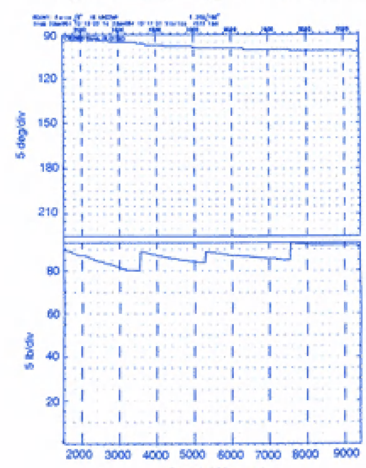


Figure 3
Force-Phase plot

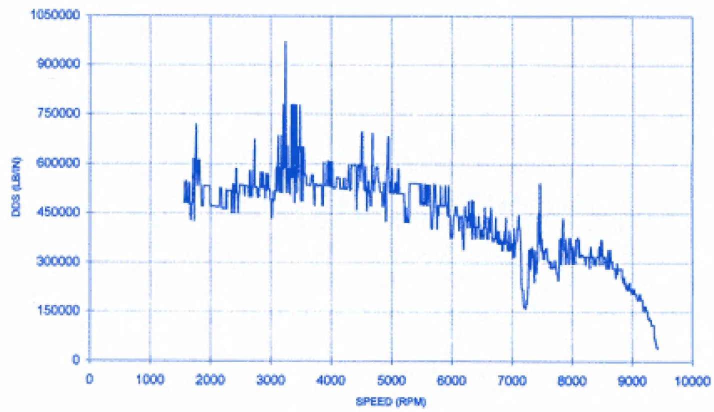


Figure 4
Direct Dynamic Stiffness graph

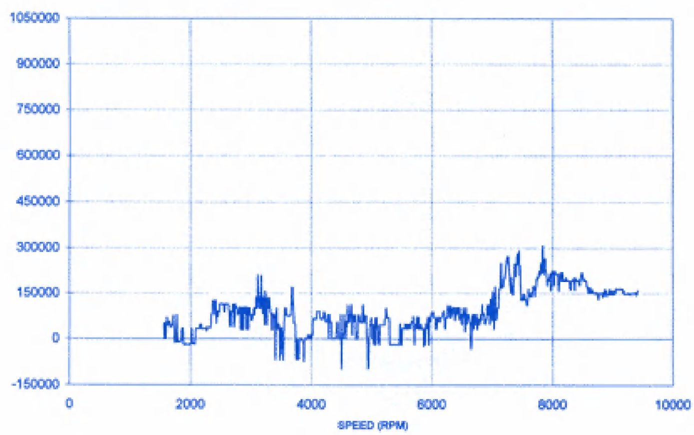
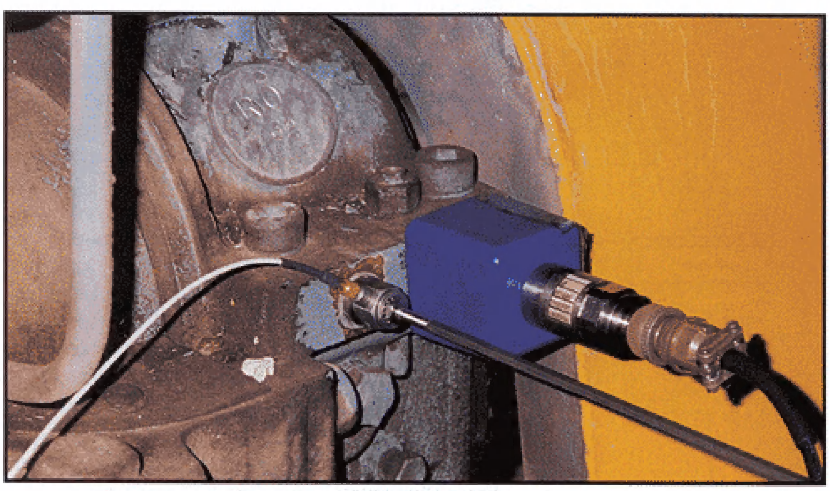


Figure 5
Quadrature Dynamic Stiffness graph



Force Transducer and Bently Nevada Velomitor® mounted with a Super Mag.

In essence, each transducer is "calibrated" so a force measurement can be inferred. For assessing machine operating integrity, these force measurements provide us with considerably more accurate information than does the housing velocity measurement alone.

Consider two machines with entirely different bearing housing support structures. Both operate at 3600 rpm. Machine A, with a compliant bearing support, has a DDS of 150,000 lb/in. Machine B, with a stiffer bearing housing support structure, has a DDS of 570,000 lb/in. Assume a velocity transducer, mounted vertically on a bearing housing on each machine, measures 0.3 inches/second zero to peak . First, we integrate the velocity measurement to displacement:

$$2X \ \frac{0.3 \text{ in.}}{\text{second}} \times \frac{1}{2\pi \ (60 \text{ Hz})} \times \frac{1000 \text{ mils}}{\text{inch}}$$

$= 1.59$ mils pp

Now we can calculate the Apparent Dynamic Force at each bearing, according to the formula:

Apparent Dynamic Force
= (Bearing housing DDS) × (Bearing housing displacement, zero to peak)

For machine A, we calculate:

Apparent Dynamic Force = (150,000 lb/in) × (0.00079 in zero to peak) = 118 lb zero to peak

For machine B, we calculate:

Apparent Dynamic Force = (570,000 lb/in) × (0.00079 in zero to peak) = 450 lb zero to peak

The same bearing housing velocity measurement on two different machines yielded force calculations that differed by a factor of almost four. Remember that dynamic force is a consequence of the shaking forces generated by mechanical defects. If the two machines had rolling element bearings, it would be obvious that machine A was in better operating condition.

We can carry this analysis one step further by considering both the static and dynamic forces acting on the bearing. Minimizing the dynamic force can maximize the life of the bearing subjected to a specific static load. Knowing

this, and the bearing manufacturer's specifications, we can approximate the life of the bearing. In the case of machine B above, the bearing may be overloaded and its failure imminent.

By determining dynamic force, this technique makes best use of housing-mounted transducers. Dynamic force, resulting from vibration, causes bearing failure. The machine's condition cannot be accurately assessed without considering it. On machines with rolling element bearings, generic vibration severity charts are not good tools for estimating acceptable vibration levels, because they don't account for housing compliancy, and, therefore, dynamic force.

Field test

We field-tested this technique on four different pumps, of either in-line, between-bearing or overhung configuration. Two of the pumps were operating and two were not. Our intent was to "calibrate" true vertical and true horizontal motion at each bearing on each pump, generate the DDS and QDS curves, and calculate the apparent force at the frequencies of interest for each point. Table 1 lists the pumps.

We used a 100 lb peak to peak electromagnetic shaker as the excitation source. A hoist supported the shaker near each test point. A 5/16 inch diameter, 12 inch long "stinger" coupled the shaker to a force transducer. The force transducer was epoxied directly to the bearing housing. A function generator, with sine wave output, controlled the excitation force amplitude and frequency. The function generator's zero crossing point was used as a Keyphasor® trigger. We maintained the shaker's output force at approximately 90 lb pp while we swept the excitation frequency from 200 cpm to as high as 13,000 cpm. We measured bearing housing response with magnetically- and epoxy-mounted Velomitor® sensors. The data was recorded on a Bently Nevada ADRE® for Windows System.

The resulting data was processed in a spreadsheet program that generated the DDS and QDS curves. Figure 2 is a Bode plot generated by ADRE for Windows during a typical calibration run. Figure 3 is the accompanying force-phase plot. Figures 4 and 5 are the DDS and QDS graphs generated from that data. We limit discussion here to the DDS curve, since our main interest is the maximum force transmitted at operating speed.

The DDS curve is constructed by plotting the DDS from each point against the operating speed at that point. Generally, a parabola is formed. If the parabola is extrapolated to the left, its intersection with the vertical axis is the Direct Dynamic Stiffness at zero excitation frequency. The point where the parabola intersects the horizontal axis is the balance resonance frequency. The glitch at operating speed is caused by a beat frequency that develops when the perturbation speed matches the machine's speed. At that point, there are two sources of excitation: the shaker and the machine's vibration at operating speed. The beat frequency disturbance occurs as the shaker excitation frequency approaches and passes through the operating frequency of the machine.

The plots shown are from testing done on pump A (Table 1), a 300 hp overhung impeller pump. At 3600 rpm, its DDS is approximately 525,000 lb/in. In this example, we measured 0.8 mils zero to peak of 1X vibration on the bearing housing, and we estimated that the rotor weighed 300 lb. The Apparent Dynamic Force transmitted to the bearing is:

Apparent Dynamic Force = (525,000 lb/in) × (0.008 in zero to peak) = 420 lb zero to peak

Assume the inboard bearing supports approximately one and one half times the rotor weight. When there are no significant misalignment or process loads, the dynamic load is approximately equal to the static load. The static and dynamic loading, along with the bearing manufacturer's specifications, are the information we need to assess the machine's bearing life.

Conclusions

A rotating machine's vibration response is dependent on its Static and Dynamic Stiffnesses. If we know a machine's Dynamic Stiffness and its vibration response, we can estimate the dynamic forces that caused the vibration. This gives us a greater amount of information on which to base our decisions.

For instance, the Dynamic Stiffness can be useful as a troubleshooting tool. It can help us determine if the source of a developing problem is internal or external to the machine. Housing vibration amplitude can increase due either to an increase in force or a decrease in the Dynamic Stiffness. If vibration increased and we determined that the machine's stiffness decreased, we would suspect problems with the support structure and foundation rather than the rotor.

Our method does not estimate the machine's static forces, which are also a significant part of the machine's operation. However, this method for estimating dynamic forces at any frequency range can help us assess machinery operating condition. ■

References

1. Muszynska, A., Bently, D.E., Frequency Swept Rotating Input Perturbation Techniques and Identification of the Fluid Force Models in Rotor/Bearing Seal Systems and Fluid Handling Machines, Journal of Sound and Vibration, v. 143, No. 1, pp 103-124, 1990.
2. Ibid.

A	Dbl. suction overhung	300 hp 3600 rpm
B	Between brg. multi-stage	200 hp 3600 rpm
C	Single suction overhung	15 hp 3600 rpm
D	In-line vertical	15 hp 3600 rpm

Table 1